

COMPRESSOR DRIVE TURBINES OF HIGH EFFICIENCY AND GREAT OPERATIONAL SAFETY

by

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Galley 2-1

ABSTRACT

Steam turbines employed as mechanical drivers require special design features. The problems confronting a turbine designer are blade and shaft vibrations, bending and centrifugal stresses, torsional analyses as well as optimal designing with respect to the requirement of attaining high efficiencies. With the help of advanced technologies, the steam turbine industry has reached a level where both reliability and high efficiencies are achieved. Better sealing properties achieved through the use of shrouded bladings with sealing strips on stator and rotor rows, as well as perfect damping of the blades by means of individual shrouding members fitted to the shorter blades and damping wires mounted on the longer blades are among some of the outstanding improvements made to achieve these goals. These changes have been incorporated in more than 120 turbines which are in operation and not a single blade failure has yet been reported. The use of guide-blade carriers is favourable because of their high thermoelasticity, a circumstance that allows for fast load changes without endangering the blading with regard to radial and axial contacting. A quick replacement with spare parts is also a remarkable advantage.

INTRODUCTION

For many years, steam turbines were mostly used for power generation. Today they are also increasingly employed in industry, especially in the chemical industry as compressor, pump and fan drivers. Small generator drive turbines up to about 20 MW as well as mechanical drive turbines which are suited for very low speeds can be optimally designed in speed according to the prevailing thermodynamic conditions, because these turbines are generally equipped with gear units to reduce the turbine speed to that of the generator to achieve high efficiencies.

This is different from turbines which are employed for driving rotating compressors, because here a direct drive with-

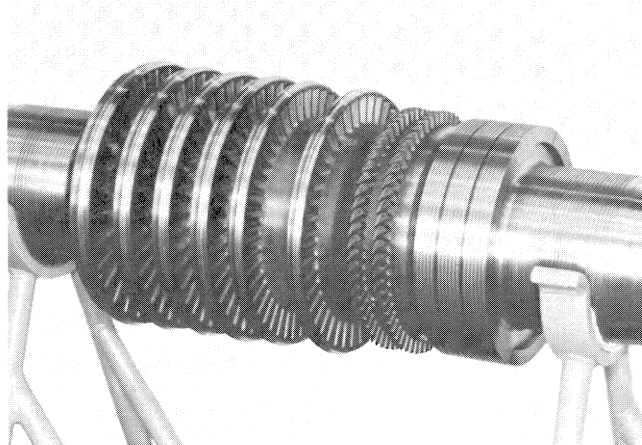


Figure 1. Rotor of a modern syngas drive turbine, type KR 25 with vibration proof, damped shrouded blading design.

Design datas: live steam conditions	115 bar/540°C
back pressure	40 bar
max. cont. speed	approx. 14,000 rpm
max. steam flow	approx. 300 t/h
turbine output	approx. 18,000 KW

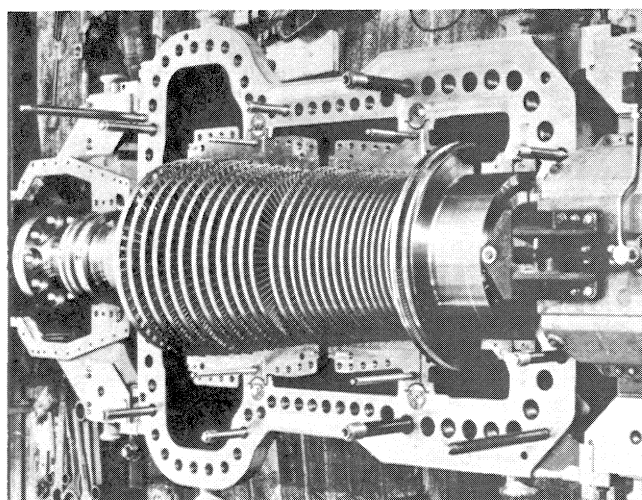


Figure 2. Rotor and lower casing part of a generator drive turbine G 50, supplied for a large desalination plant. Shroudings on rotor and stator blades. Two guide blade carrier halves. Turbine speed 3,000 rpm. Turbine output approx. 40,000 KW.

out a gear unit in between is preferred. In this case the driver has to act as a very flexible module for speeds, outputs and steam conditions. For years industry has been in a position to offer turbines of high operational safety. The inlet steam conditions of these turbines extend up to 140 bar and 540°C. They are in demand partly as backpressure turbines or as condensing machines with or without extraction, with speeds ranging from 3,000 to 15,000 rpm as well as outputs between 2 MW and 20 MW with high-speed units and up to 85 MW at approximately 3,000 or 4,000 rpm.

In the years before the oil crisis, turbine users were less interested in the steam consumption and in obtaining peak efficiencies than in achieving absolute operational safety. They were doubtlessly right — did not a turbine failure often bring a whole refinery to a standstill? Such damage would, within a few days, nullify the gain obtained through the heightened turbine efficiency. In earlier turbine designs, a high efficiency was only possible by sacrificing part of their operational safety.

BLADE DESIGN

Reaction-type turbines built in the years before the oil crisis were preferably equipped with free-standing blades.

The problems to select such types of blades to simultaneously meet the design criteria regarding efficiency, stresses and resonance as well as the occurring blade vibrations, are well known and have been more or less resolved by the turbine industry. Just think of the great effort and skill demanded from the individual turbine specialist who had to design the blading of a condensing turbine with a wide speed range in such a way that no blade resonances occurred.

Very often it became necessary to use thinner profiles in order to avoid blade resonances within the speed range. This, however, resulted in higher bending stresses. The other possibility consisted in a change of the actual blade length which, in fact, could have an unfavourable influence on the efficiency.

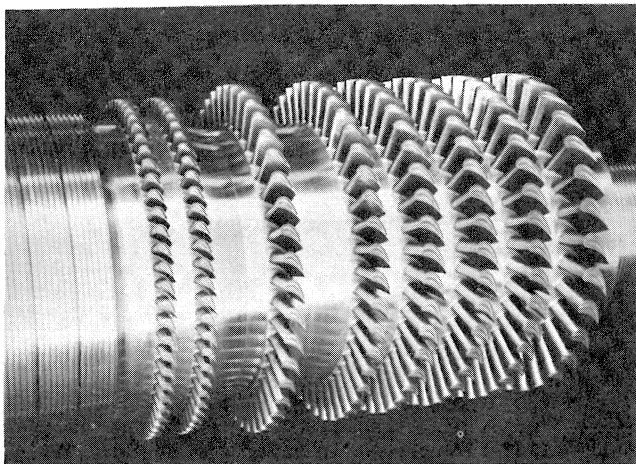


Figure 3. Rotor of a syngas drive turbine with an output of 14,000 KW at a speed of approx. 14,000 rpm. Steam conditions as mentioned under Figure 1. Contrary to the blades on the rotor under Figure 1, these blades are electromechanically machined and integral with the rotor. Although there is no need of a special fitting of the blades in the rotor grooves, these blades are more endangered by resonance compared with other blades having roots inserted in grooves and dampening shroudings.

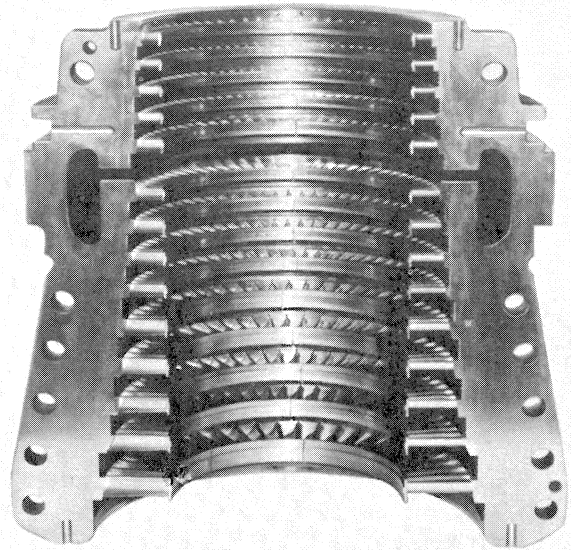


Figure 4. Guide blade carrier with shrouded blading design. They offer the following advantages:

- Speedy temperature balance between the rotor and the guide blade carrier which is in contact with steam on all sides.
- Large temperature gradient, insensitive to temperature and load variations.
- Movement in the casing partition joint does not affect the radial blade clearance.
- Easy servicing, simple and problem-free erection, quick service due to reasonably sized components.
- Quick replacement with a spare rotor and guide blade carrier in case of a failure.

These problems have been solved by employing stationary blades with riveted shrouds and rotor blades with milled-on shrouding members in cases where centrifugal forces allow their application. By taking these measures, the problems of providing a resonance-free blade design has been perfectly solved.

These types of moving blades, consisting of profiles of a high moment of resistance, combined with the said shrouding member system and fitted to the rotor slots in the specified way, show no resonance phenomena. Even at frequencies of excitation corresponding to the theoretical value of vibration, no alternating stresses were detected. Thus the method of friction damping occurring between the individual shrouding members has turned out to be very successful. Presently, more than 120 turbines are in operation with this special blade design and no blade failure has been reported at all.

Test measurements on stationary blades, that are equipped with shrouds riveted to groups of blades, have shown that the combined effect of friction damping in the rivet pins, slot damping in the guide blade carrier, as well as a small percentage of aerodynamic damping, fully suffices to keep the alternating stresses so low with vibration resonances that continuous operation is possible, even if the turbine runs in the resonance.

With the longer blades used within the condensing range, it is no longer possible to mill shrouds onto the blades due to

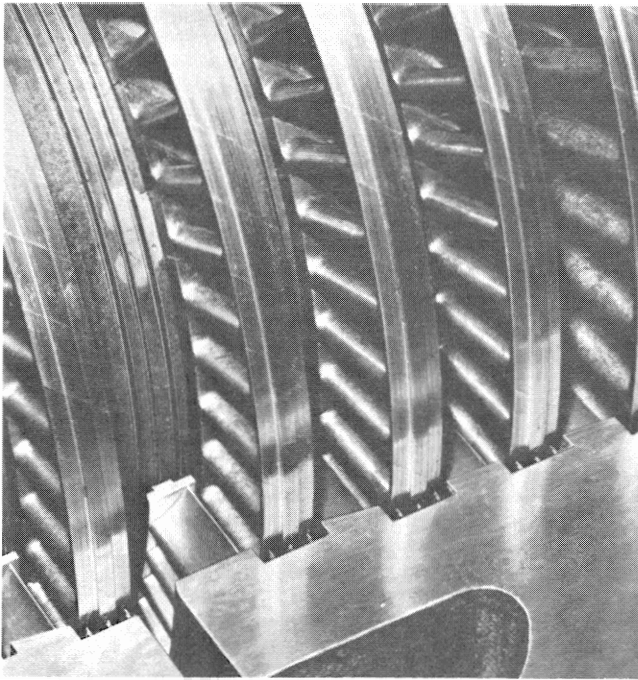


Figure 5. Heavy duty high pressure blading with shroudings and 3 sealing strips in standard design.

Tapered T-headed root

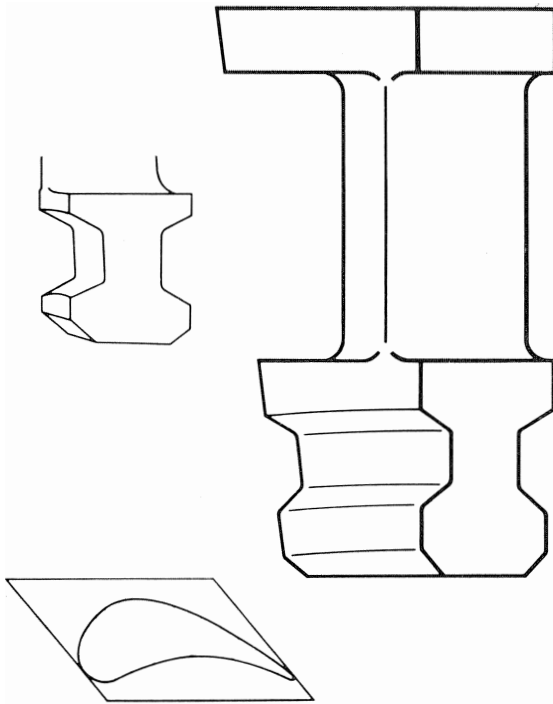


Figure 6. Blades, milled from the solid with shrouding members. For better centering and to reduce the notch factors, tapered shoulders on the roots have been provided. The centers of gravity of root, blade and shrouding member are situated on one line.

the effect of centrifugal forces. The tip to tip distance of long blades would make the shrouds too heavy, and there would be the danger of overstressing both the blade profile and the blade root fastening by the effect of the centrifugal forces.

Here damping wires are inserted which are loosely placed into the well-rounded holes of the rotor blades. The fitting height of these holes in the blade is dictated by its specific shape and thus by the position of the natural frequencies due to design.

The diameter of the wire is then only a function of the maximum speed, of the wire material, and of the blade pitch at the level of the damping wire holes. We use solid and hollow steel wires or titanium wires, as per requirement. These types of techniques used to reduce the vibration levels have proved to be very successful in hundreds of turbines.

Compressor drive turbines, when accelerating a process unit, will often have to be operated over wide speed ranges, for which they were not designed originally. In this case only the damping wires, as mentioned before, can help to assure an operational safety.

It seems to be necessary, however, to consider the overall stress level of the blading, in order to avoid bending stresses which would be too high at full load operation condition. Operationally-safe blades operating under full-load conditions will have bending stresses that, within the superheated steam zone, provide a safety margin 8 to 10 times greater compared with their yield point.

Stresses caused by centrifugal forces shall not exceed $\frac{1}{3}$ of the yield point at maximum speed. These days, 13% chromium

Damping effect

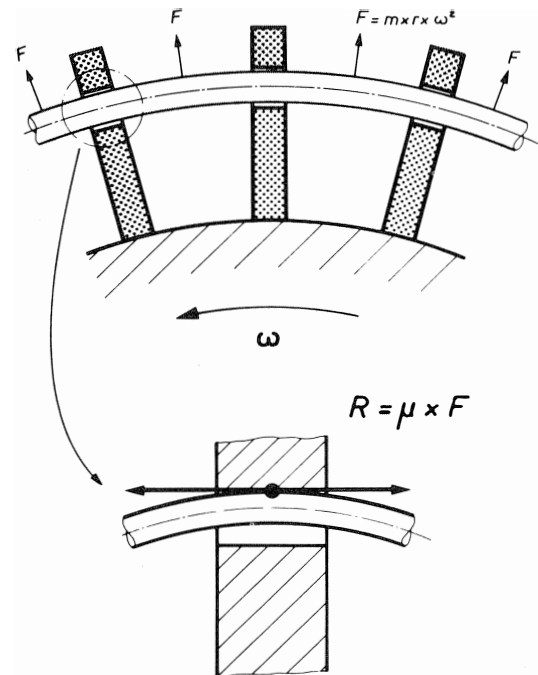


Figure 7. Schematic of a blade with inserted damping wire which due to the influence of the centrifugal force exerts a friction force onto the vibrating blade.

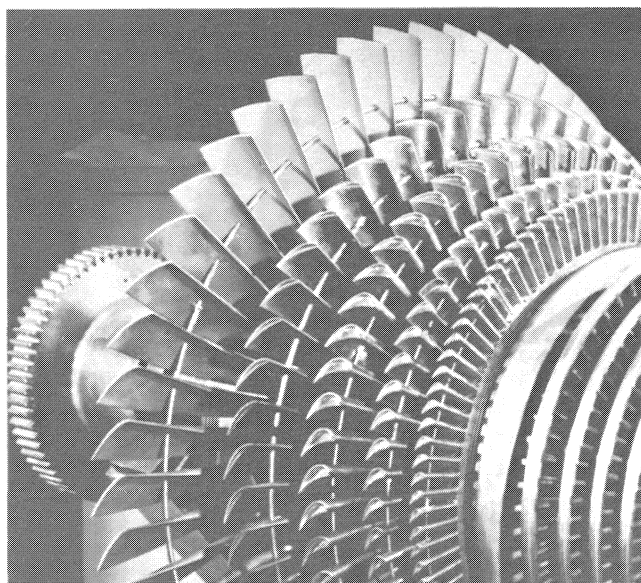


Figure 8. Low pressure part of a condensing turbine as compressor driver with free-standing blades equipped with damping wires to achieve a vibration proof design.

steels are usually employed as turbine-blade materials. The use of this steel is based on many years of operational experience, even under the influence of corrosive media. An inferior steam quality is certainly not of benefit, especially for the blades of a condensing unit, but also under these aggravating conditions, the user can achieve an acceptable life span of the blading by using blades of great chord length with especially low static stress values. This method is used above all in the transition zone between the superheated and wet steam areas and in the wet steam zone itself, so that the above safety factors for the bending stresses are doubled.

BALANCING

Another factor worth mentioning is the design of the turbine shaft itself and its susceptibility to shaft vibrations due to inevitable imbalances caused by blade deposits or by a displacement of the coupling during operation.

When considering the measured stiffness and damping values of the journal bearing used, it becomes possible with the aid of modern computers to predetermine most realistically the vibration amplitudes occurring in the bearings and on any other spot along the shaft are calculated.

The maximum permissible balancing quality is used as a basis, specified normally as $Q = 2.5$ (mm/s), with which the vibration amplitudes occurring in the bearings and on any other spot along the shaft are calculated.

In practice, however, balancing qualities better than $Q = 1.5$ (mm/s) are obtained so that the amplitudes occurring in the bearings during the running test are reduced proportionally as against the precalculated values resulting in great reserves.

Quiet running of the turbines, however, requires that the rotor with its blades and the overspeed trip bolt, etc., are of well balanced design. In this connection, I would like to mention the favourable arrangement of the blade locking segments in the individual blade grooves across the rotor.

Assumed imbalances

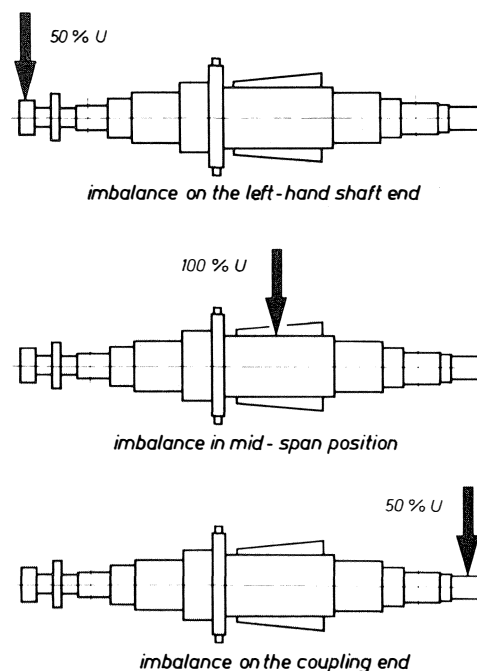
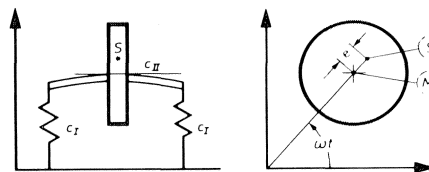


Figure 9. Assumed imbalances on which the computer calculations of the shaft vibration behaviour are based.

Shaft Eccentricity e Balancing Quality Q

S = centre of gravity

M = geometric centre



c_1 = Stiffness of the bearings

c_{II} = Stiffness of the turbine shaft

e = Shaft eccentricity

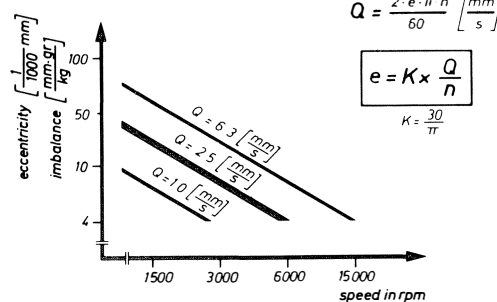


Figure 10. Relation between balancing quality Q , turbine shaft speed n and permissible shaft eccentricity e , demonstrated by means of an unbalanced shaft supported in journal bearings.

Results of balancing procedures on steam turbines in the AEG-Kanis factories

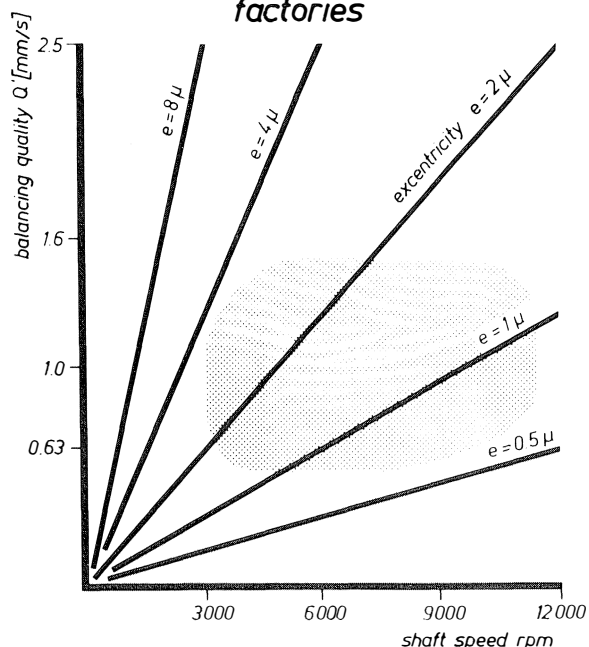


Figure 11. Range of results of the shaft balancing qualities achieved on our test beds. This chart shows the calculated balancing qualities of about 80 turbines after being balanced.

Shrouded rotor row with sealing strips

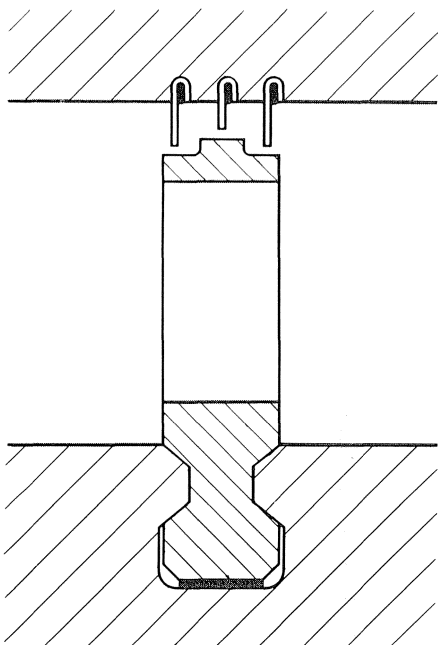


Figure 12. Typical rotor blade with shrouding member milled from the solid. In order to achieve a tight fit of the blades in the grooves, a small steel band is inserted between the groove-bottom and the blading.

Moment-balancing of the large final blades was a great success. Since these blades are either copy-milled or forged, there are greater variations in weight than with cylindrical shrouded blades. The eccentricities occurring when the blades are fitted in random arrangement can be strongly reduced by properly distributing the weighted blades along the circumference after calculating their position by computer.

Further, the blades must be firmly fitted to the rotor to avoid possible loosening during operation, for this would certainly entail a change in the state of balance.

The tight fit of the rotor blades in the shaft slots is achieved by inserting a steel strip underneath the inverted T-roots and fir-tree roots in the rotor slots. The finger-type roots used in turbines subjected to high centrifugal stresses are tightly fitted to the shaft by means of a special pin-fastening.

pin fixed 4-lobed finger-type root

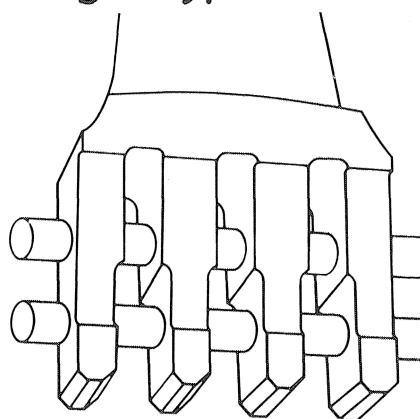


Figure 13. Design of the final stage root of a condensing type turbine for high centrifugal forces. Depending on the prevailing stress level the roots are designed with up to 6 lobes and up to 3 cylindrical pins.

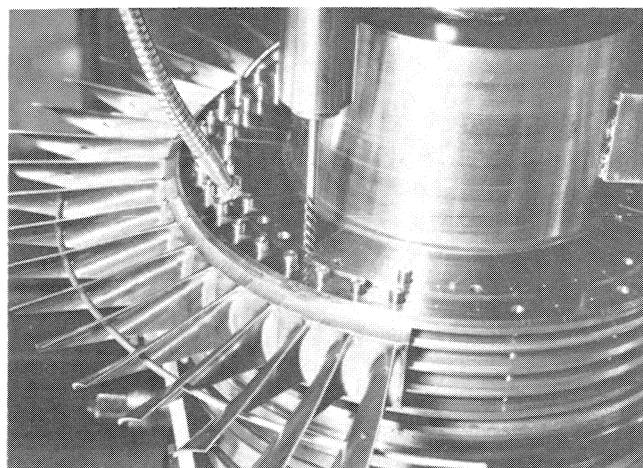


Figure 14. Procedure of final stage fastening to the rotor with multi-lobe root and damping-wire. After joint boring of rotor and blade root, the holes are reamed to size. Then the pins are inserted which only need locking against axial displacement.

For the entire turbine design it has proved to be an elegant solution to accommodate the stationary blading in one or several guide-blade carriers, and not directly in the casing as in earlier designs. The measure offers a great many advantages, especially in view of obtaining a uniform thermal expansion of the guide-blade carrier and the rotor.

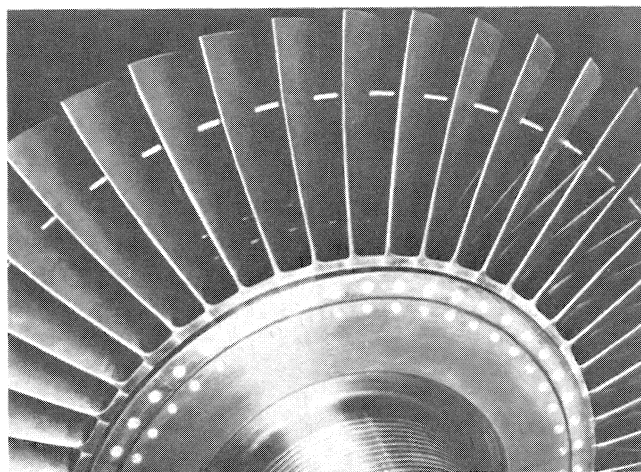


Figure 15. View in axial direction of the finished pin fixed final stage blade with multi-lobed root.

This leads to smaller permissible radial clearances, which, in turn, favourably influence the efficiency without endangering the operational safety. Also, in case of a possible damage, the guide-blade carrier and rotor can be replaced together on site within 1 or 2 days by the spare rotor and stator, without the turbine having to be returned to the manufacturer.

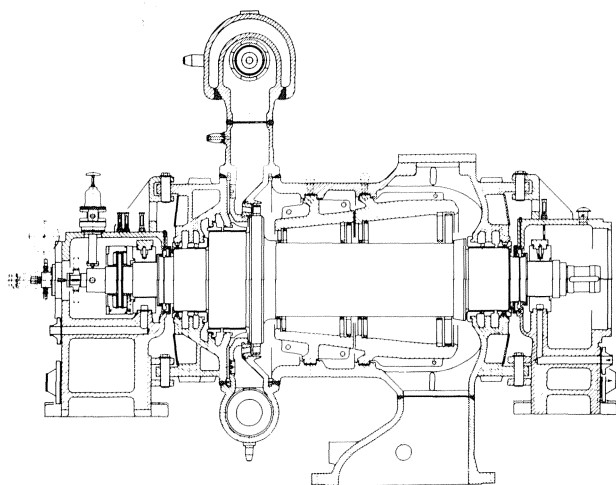


Figure 16. Sectional drawing of a backpressure turbine with two guide-blade carriers halves, installed thermoelastically in the turbine casing.

BLADE LOSSES

Shrouded blades are employed in modern turbine engineering to avoid blade vibration resonance problems. This has another advantage, namely a reduction in leakage losses on both the rotor and stator blades. These losses occur in the

reaction stages as a result of the pressure difference before and after the blade rows, in contrast to the type of blading used in impulse type turbines. These leakage losses are reduced to nearly half the value occurring with free-standing blades by fitting 3 sealing strips each opposite the rotor and stator rows. Due to the shrouding, the conveyance of steam through the blade path is much improved. Furthermore, a possible friction damage is less likely with this design than with blades having free standing edges.

The final stages in the condensing section play a considerable role in handling the turbine gradient so favourably. For this purpose, heavily twisted blades are used today which more properly adapt themselves to the different triangles of velocity at the roots and the blade tips. Further, the angles of incidence are chosen in such a way that the steam outlet from the last condensing stage is almost axial and will only be slightly influenced by the various load conditions. This is to reduce the inevitable exhaust losses to a relative minimum.

In short blades, however, all thermodynamic processes can be referred to the blade centerline, because the triangles of velocities at the root and tip are almost identical. Therefore, cylindrical blade profiles may be chosen without adversely affecting the blade efficiency.

An important factor that must not be neglected in profile shaping is the loss in efficiency in partial and overload ranges. A well-rounded leading edge — as presently employed in reaction-type turbines — will provoke a slight reduction in efficiency at an optimal oncoming flow, as against the sharpened leading edge normally used in impulse turbine construction, although the behaviour is much more favourable in the case when the machine is operated under service conditions varying from the design point. For this reason, the reac-

Efficiency of the reaction section

$$\eta_{\text{Ü-Teil}} = \eta_{\text{Blade}} \times \eta_{\text{Gap}} \times \eta_{\text{Fan}} ;$$

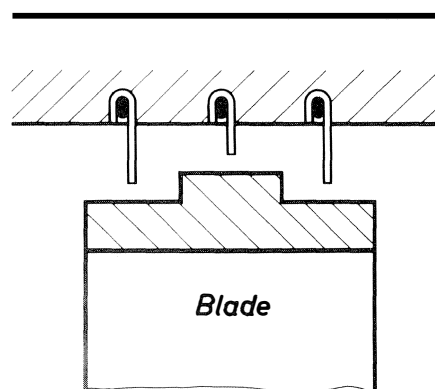


Figure 17. The turbine blade efficiency results from the product of profile efficiency, gap efficiency and fan-type efficiency.

Actually twisted blade with 4-lobed finger-type fastening

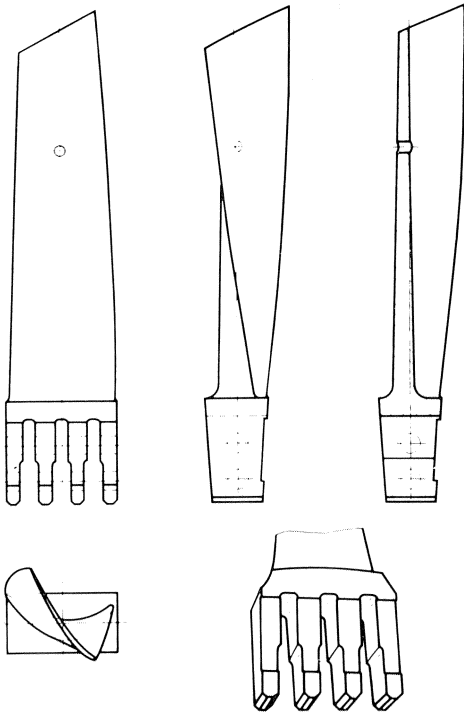


Figure 18. Drawing of a twisted blade of the final condensing stage with multi-lobed root and damping wire employed for high-speed compressor drive turbines.

tion turbine blading is superior to that of the impulse turbine especially in efficiency, even though its higher leakage losses use up part of the efficiency gain.

A comprehensive computer program has been developed, with the aid of accurate grid-type wind-tunnel tests, to allow for optimal design and calculation of steam turbines. This program can be used to predict the behaviour of a steam turbine at variable operating conditions with an accuracy within 1%, a fact that was confirmed by numerous measurements. This provides optimum adaption and design of a turbine to the individual requirements. With the aid of the computer, program changes in design can be evaluated; such as, whether a wheel-chamber bypass or an internal bypass for the higher loaded points will improve the efficiency.

The computer program for the blade steam path design uses standardized blades of a great variety of blade and chord lengths, accurately defined in the design. This program also includes any bleeding and extraction points automatically. Under this program, the blades for the stationary and moving rows are selected in such a way that the specified stress-limiting values are not exceeded; moreover, this program provides some further criteria, such as the wheel-chamber pressure desired or the maximum permissible blade length that has to be considered for a certain type of turbine.

A turbine is not a simple "component," especially if high demands are made from turbine efficiency and reliability. This is mainly the case with turbines intended for driving compressors, pumps or fans, conditioned by the variable speed ranges

Blade efficiency on reaction- and impulse-type turbines as a function of speed variation

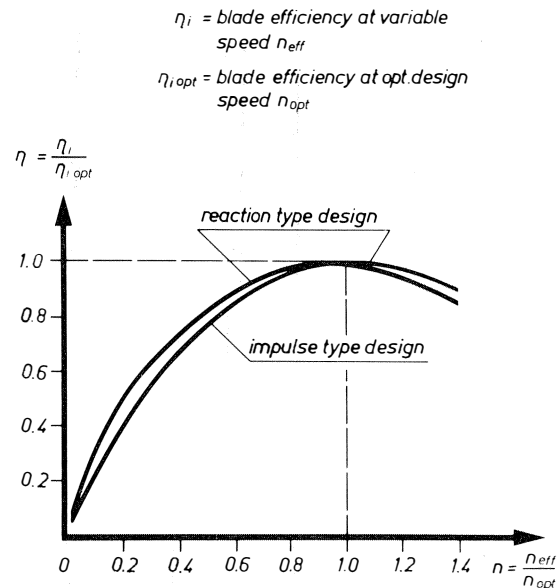


Figure 19. Difference in efficiency of impulse and reaction type turbine blading as a function of the momentary shaft speed. Compared with the impulse type turbine, the reaction type turbine offers a more favourable efficiency, especially in the field of compressor drive turbines with wide speed ranges.

and the requirement for lower running costs. With backpressure turbines of the reaction type, turbine efficiencies of more than 85% are obtainable within the range of small outputs under favourable conditions. In condensing turbines, the adiabatic efficiencies are between 80-85%. These values, however, include all moisture and exhaust losses.

TURBINE GOVERNOR

Unfortunately, when speaking of the operational safety of a turbine in general, one easily forgets to mention the governing system of the turbine and the pertaining valves, i.e., the closing devices required. To achieve a high degree of safety, hydraulic turbine governing systems are recommended.

Consisting of a few, uncomplicated components, this system fulfills all requirements that are nowadays demanded from a modern turbine governing system. The hydraulic fly-piston governor operates according to the force-balance principle, where each variation in speed is directly converted into a signal pressure (p_3) proportional to the output, which, in turn, serves as a control signal for the oil relay to regulate the stroke of the servomotor. The speed adjuster is mainly composed of a simple pressure reducing valve, the outlet pressure (p_2) of which counteracts the piston centrifugal force of the governor.

This governor has the advantage of a very high, not actually measurable response sensitivity ranging from less than 0.01% up to 0.1% of the nominal speed and it has also an excellent dynamic behaviour with a time constant of only 15 ms.

This speed governor has an operating range between 0 and 110% of the rated speed, permitting thus a controlled startup from standstill which is especially required by automated units. Furthermore, it is possible to equip the governor with a pneumatic, hydraulic or electronic remote speed adjuster, which proves that this governor is also excellently suited for fully automated operation.

The centrifugal governor acts upon the valves via the lift-servomotor. As a standard, we use consecutive compound valves, which are arranged in series in a casing above the turbine. We normally use our double seated valves for all our turbines. Because of their symmetrical working surfaces for the inlet steam and nozzle pressures which require only small operating forces, they also need only small, quick-acting servomotors. The valve 1 is directly controlled by the servomotor. When the maximum flow is attained in the first nozzle group, the second valve, loaded in closing direction by a spring, is opened via a spigot on valve 1. The third valve is then opened when the second valve is fully open.

The individual valve spindles are concentrically supported on both sides at a short distance in the valve cage. Hardened teeth safeguard the valves against vibrational rotation and thus against excessive wear. The valve cages which are easily re-bleed off, which, in addition further improves the efficiency.

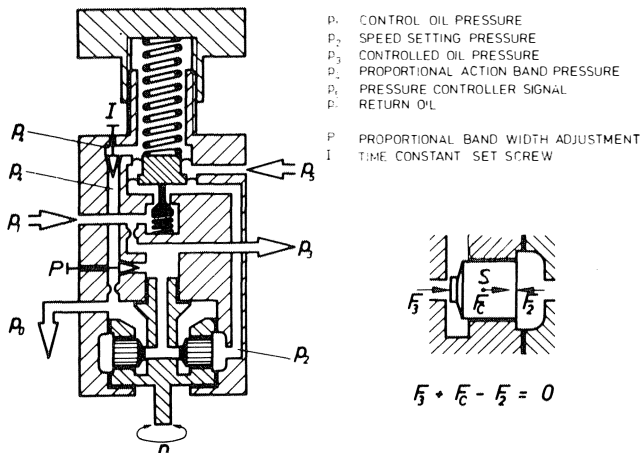


Figure 20. Sectional drawing of the fully hydraulic fly piston governor which, contrary to other centrifugal force operated governors, does not require the force of a spring or membrane as the spring action of a rebound nozzle is employed. Thus a lift of less than 0.2 mm suffices to cover the entire operating range.

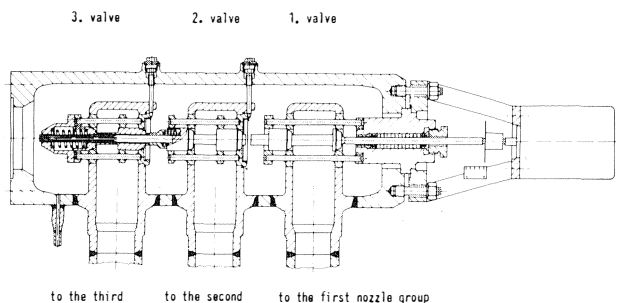


Figure 21. Valve block incorporating 3 control valves. They are opened consecutively by means of a hydraulically operated valve spindle in axial direction.

movable are so designed that their resistance to the steam flow is minimal and their thermal expansion coincides with that of the valve spindle. By this means, good sealing characteristics are obtained.

All the valves are operated by the valve spindle. The spindle is sealed with a packing gland, so that no gland steam has to

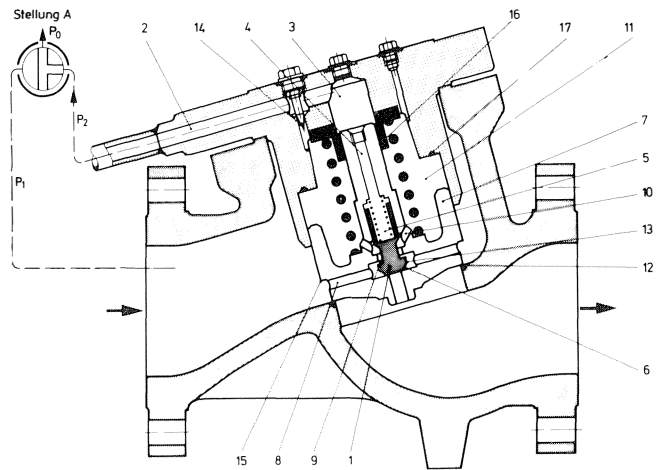


Figure 22. Sectional drawing of a steam-operated, hermetically sealed trip-valve. The valve is shown still in closed position while the 3-way control valve has just been switched to the opening set point.

This direct drive by means of a driving rod, which acts directly from the power piston upon the valves, does not require any mechanical deflecting equipment, such as levers or similar devices which may seize or tend to develop wear symptoms. Hence, only 5 moving parts are needed for 3 valves, these being the 3 valve spindles and the 2 reset springs.

Emergency trip valves are used as main isolating devices for the steam leaving the boiler and entering the turbine. For this purpose, we mostly employ newly developed steam-operated valves that distinguish themselves by their very short closing time.

These valves are equipped with a locking system by means of a pilot-piston, which opens the emergency trip valve only if a pressure of about 80% of the initial pressure has built up between the closed control valves and the emergency trip valve. This avoids the danger of any damage due to thermal overloading. These valves need no outside valve actuators. They are opened by the steam pressure itself. To initiate the control action, a small 3-way valve will be provided, which may be actuated as preferred, either by hand, hydraulically, pneumatically or even electrically.

It goes without saying that due to the low number of moving parts involved, a high operational safety is obtained.

There are no mechanical passages to atmosphere. For this reason, continuous supervision and maintenance of the packing gland leak-off become superfluous. This valve is completely steam-tight both in open and closed position. Thus no gland steam develops and no salt deposits will form on the moving parts with their detrimental effects on turbine operation. For the same reasons, these valves are also suitable for application in nuclear power stations; this is what they were recommended and approved for.

FUTURE DEVELOPMENT PROGRAMS

The following are a few focal points concerning the future development of turbines and their further improvement.

Advance stress calculations, using the methods of "Finite Elements" are a very important part of developing high reliability turbines. To achieve higher wheel chamber conditions at low material expenditures without lowering operational safety, optimization of turbine casing joints must be achieved. Figure

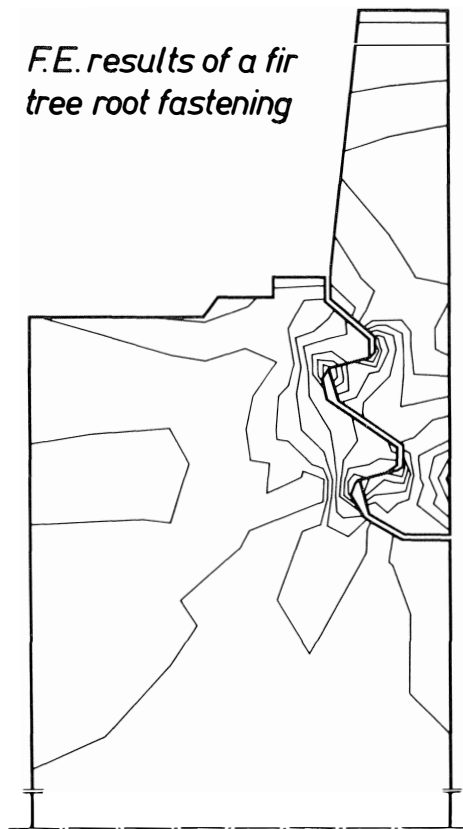


Figure 23. Finite element results of a fir-tree root fastening.

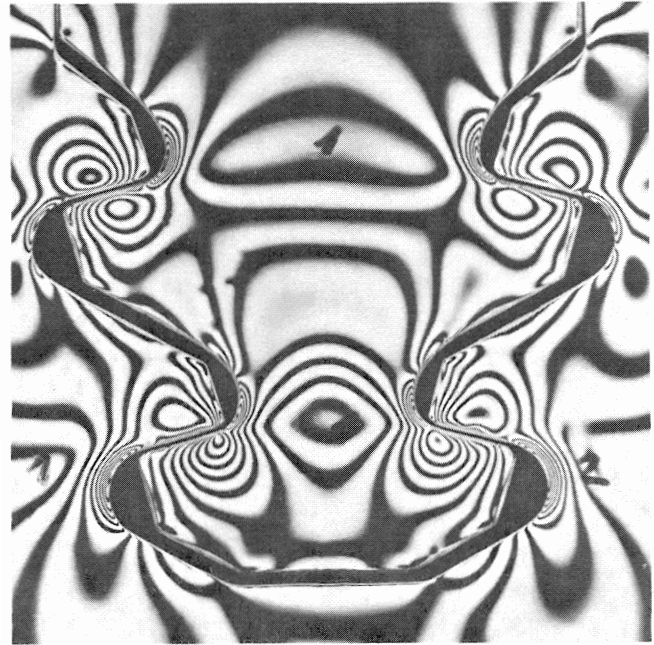


Figure 24. Stress analysis of a fir-tree root by means of the stress and strain method. The number of rings indicates the stress level prevailing on this very spot.

23 shows the results of the finite element program as applied to a fir-tree root. Photoelastic results of the fir-tree, as seen in Figure 24, show excellent agreement.

Oil whip studies in journal bearings require closer investigation. As more research is done in this area, better control of this phenomenon will be achieved.

Optimization of the start-up process is required since, during start-up, relative expansion and thermal stresses play a dominating role. Machine life counters are being developed to add up thermal shocks and rough load variations according to the material stresses occurring. This counter could be very helpful in indicating problems of thermal fatigues and could indicate, for instance, when casing joint bolts have to be replaced.

